SIMULAR,[1] INC.

Mechanical Design in a Radar Simulator

An electronic radar simulator being developed by the Simular Corporation,[1] Los Angeles, California was to include a moving "window frame" 26 inches high by 50 inches wide weighing over 150 pounds. This frame, which was about two inches thick and stood in a vertical plane, had to be able to move 20 inches up and down and 20 inches from side to side (10 inches each way from center) past two optical scanning lenses. To provide movement speed of 1/2 inch per minute two servo motors of four watts each were geared to the frame through lead screws For optimum performance of the servos it was necessary that they encounter approximately the same load in either direction of frame movement. Mr. Irv Howard, chief mechanical engineer at Simular was concerned about the weight of the frame. In particular, he wanted to be sure that it would be adequately supported to be durable and hold the required tolerances, and that the servo motors would not be unduly loaded.

[1] Names in this case are disguised.

Prepared in the Design Division of the Department of Mechanical Engineering by Karl H. Vesper with financial support from the National Science Foundation.

Basic Principles of the Simulator

The simulation process in the machine began with two optical lenses scanning two glass plates on which were drawn images of terrain which the radar was to "see." The image on each plate (costing up to a million dollars to make, scale 1 inch to 100 miles) looked somewhat like a photographic negative of the earth seen from high above. Each lens would look through the negative toward a light shining from the opposite side and producing a "spot" of diameter 0.001 inch. As the plate moved between lens and light the dark and light regions of the negative would cause variations in the amount of light seen by the lens and correspondingly affect what appeared on the radar screen viewed by a pilot trainee sitting nearby in a simulated airplane cockpit. Depth of field for the lens to maintain resolution required that the distance between plate and lens be held within 0.005 inches of nominal.

When a radar views real objects it responds to two properties of them, one being the perpendicularity of their surfaces relative to its line of sight and the other being the reflectivity of the material from which they are made. To impose these two types of effects the simulator used two plates, one for each effect, mounted side by side as in a stereo viewer, each with its own scanning lens and light. Signals from the two lenses were fed first to a computer wherein they were coordinated and then to the radar output and screen.

To operate properly the two pictures had to be positioned accurately relative to each other (within a tolerance of approximately .0001 inch). The size of each plate was to be 20 by 20 inches, and each plate was to be located in a vertical plane (like a house window) between its lens and light source, these being 18 inches apart. In moving past the lenses to permit scanning, a maximum speed of 1/2 inch per minute was needed and a translation accuracy in the plane of movement had to be held within 0.001 inches. This made it important, in Mr. Howard's view, that no "backlash" be allowed in the mechanical system for moving the plate.

The general mechanical arrangement he expected would consist of a fixed frame plus two moving frames. On the fixed frame would be a horizontal support shaft and a guide along which the horizontal frame would slide horizontally. On the horizontal frame would be a vertical support shaft and a guide along which the second moving frame containing the film plates could slide vertically. Thus the film plates could be moved both horizontally and vertically by electric servo motors acting through lead screws. He commented that no serious consideration had been given to laying the plates flat horizontally, partly because this would take more floorspace for the machine and partly because the plates then might sag out of tolerance. Both these considerations in his view were important.

Background of the Project

The simulator project, which was now supported by a government contract calling for delivery of four machines, had begun in the Proposal Section of the company. A team of engineers including

Mr. Howard had prepared a proposal in response to a government request for quotation The expectation at Simular was that if the first four machines worked well there would be orders for others to follow.

During initial discussions in the proposal team Mr. Howard "roughed out" an overall sketch of the mechanical part of the machine, a copy of which appears in Exhibit 1. He expected from previous experience on similar machines that maintenance of tolerances would be important and that to lighten servo motor load it would be desirable to provide the moving frame with a counterbalance. For the latter he expected to leave free space at one end of the frame. However, to keep the floorspace requirement of the machine to a minimum he thought it desirable to shape and mount the counterbalance as compactly as practically possible.

Mr. Howard, a graduate mechanical engineer, had been with Simular for five years following three years as a mechanical engineer in a large aircraft company. He commented that although he worked exclusively on mechanical engineering problems the company was primarily concerned with electronics. "There are plenty of mechanical problems on machines like this," he commented, "but the mechanical engineer is seldom in charge. In an electronics company it's usually an electrical engineer who leads as project engineer. There are about 35 electrical engineers in the company and only 4 mechanical engineers.

"The mechanical engineer who does work like this has to be a generalist, ready to take on any sort of problem. One minute I'll be worrying about mechanisms and the next I may be on structures, shock and vibration, or air-conditioning problems. We've had specialists on subjects like stress analysis apply for jobs here but have never hired any. The three graduate M.E.'s under me have broad experience. They're good men because no matter what comes up, they can get the job done."

Mechanical Design

Mr. Howard's approach to the radar simulator design was first to sketch roughly the overall framework for mounting and moving the film plates and then to concentrate on mounting them. Around both of the two 9 pound plates of 1/4-inch-thick glass he planned to fasten with rubber potting compound an aluminum support frame weighing 15 pounds. This frame in turn would be fastened by thumb screws into a larger frame of cast aluminum to run on "ball bushings" (illustrated by catalog excerpt in Exhibit 2) along the vertical support shaft at one end with power from the vertical servo. A rail and roller guide at the opposite end of the frame would prevent rotation about the vertical shaft but would not provide any support. His estimate for the weight of the larger cast aluminum frame and its bushings was 134 pounds, so total weight with the two plates mounted would be 167 pounds.

"To estimate the weight of the frame," he said, "I first drew it so it looked stiff enough. Then from the sketch I estimated the volume of metal and multiplied by its density. ... By spending more time I could probably have made the frame lighter. I could have made one, applied loads and measured deflections and kept reducing the metal until it was

no stiffer than it had to be and as light as possible. But I'm not a stress specialist, I'm a generalist. Also, this is not a mass production of the design. The main consideration was holding tolerance on the film plates. For the scanning lens to operate properly there should not be more than a half thousandth play due to deflection, and this argues for making the frame and supports very rigid. Weight, on the other hand, is not so important in this machine, since it will be just sitting in a room, not flying in a plane. I have seen an engineer spend days refining for an ounce when a pound would do, and that, in my view, is a mistake."

Having roughed out the weight and dimensions of the frame, Mr. Howard thought he next should work on details of mounting it. In particular, he thought the next two components to concentrate on should be (1) the vertical support shaft and (2) a counterbalance for the weight of the frame. His main question for the vertical shaft was what the cross-sectional dimensions should be. For the counterbalance he had decided to use a metal can filled with lead shot and hung by a cable over pulleys from the frame. The questions here as he saw them were: (1) what should be the dimensions of the counterweight can to hold the lead shot, (2) where should the counterweight cable be attached to the frame holding the film plates and (3) what size cable should be used?

Evolution of the Counterweight Idea

Mr Howard had thought of using a spring but he quickly saw a difficulty, in that the spring tension would vary as it was stretched, causing the counterbalance action to be too weak at one end of the travel and two strong at the other end. He recalled that there was a type of spring called a "negator" which exerts constant tension as it is stretched. But then, he reasoned, someone might lay the machine on its side, and there would be nothing to counteract the spring action. "Anyway," he said, "large negator springs are hard to get and hard to use. For this application it would have to be very large, and it would probably be dangerous to install. A thing like that could cut your fingers off."

"From the beginning of this design I assumed we would use a simple counterweight. I had needed one several years ago where we were using a little motor to move a heavy object vertically, and I've used them in a couple of designs since then, so the idea occurred easily. When I laid out the basic framework of this machine I left room at one end for the weight to hang."

A sketch showing the location of the rectangular space approximately 6 by 7 inches horizontally and 52 inches vertically left free for the weight appears in Exhibit 3. Mr Howard considered the 7-inch dimension fixed by mounting constraints of the lights and lenses. To make the weight fit compactly in the machine he assumed it would be rectangular in shape except for a radius of 1-1/2 inches at one corner to fit the contour of 1/4-inch-thick adjacent aluminum shield. The 1-1/2 inches represented the minimum bend radius for such sheet, according to one of Mr Howard's handbooks. The 6-inch dimension was,

he thought, controllable but to be minimized, since increasing it would enlarge the floorspace required by the machine.

"As a material for the weight," he continued, "I wanted something to be heavy but not take up too much space. In my desk drawer I keep an issue of "Materials in Design Engineering" magazine which lists specific weights of different metals. The heaviest is Osmium, and below it are things like platinum and depleted uranium. The first reasonably priced material is lead, and that's what I picked."

"I thought of casting the lead in a bar, but since I wasn't sure precisely how heavy it should be I thought there should be some way to adjust the weight. So I considered using a stack of plates on a pole. It seemed simpler and more solid to make a can and pour in the right amount of melted lead. But again there was the problem of how to adjust the weight. And besides, hot lead can be poisonous. So I settled on the idea of filling a can with shot."

"Then there was the question of what volume to allow in the can. The magazine gave a specific weight for lead of .41 lb/in.^{3}, but none of my references said anything about the specific gravity of shot. I thought it would be lower for shot than for solid lead and higher the smaller the ball size. But nobody in the office even knew what sizes were available."

"So I drove a few blocks down to the Stanford Sport Shop and looked at shotgun loading supplies. They had some shot about 1/16 inch diameter. I imagined a close view of the shot piled up. I could see a sort of triangular prism of space between the balls. I just guessed that this would represent about three tenths of the total volume, and so I assumed a "space factor" of 0.7. In aircraft design where weight is terribly important it would probably be worthwhile to study something like this analytically. But for this machine a guess seemed good enough. I don't particularly take pride in being a scientist, but I do in getting the job done."

For the can to hold the shot Mr. Howard thought he would use steel sheet 1/16 inch thick on the sides and 3/8 inch thick on the top and bottom (.287 lb/in.^{3}).

Mounting the Weight

As he envisaged it, the counterweight can would have a hole through it from top to bottom to allow for a vertical guide shaft along which the weight would run. He selected a spliced shaft with a "ball spline" which required a 4-1/2-inch diameter hole in the counterweight can. To choose the shaft diameter he guessed a size and then computed the natural frequency of vibration which the shaft would have when the weight was halfway between top and bottom. This frequency calculation appears in his notes reproduced as Exhibit 4. The formula used was one which he had copied a few years earlier from a magazine article and written in the margin of a handbook.

"On a similar job some time back I learned the importance of making this calculation," he said "That job involved a piece of equipment to be installed on shipboard, and the specifications required that it be tested on a "shake table" at frequencies ranging from 5 to 30 cycles per second. We had used a much smaller guide shaft relative to the weight and its natural frequency was somewhere in that range When the shake table hit that frequency the weight started vibrating and threatened to break loose. It was terrible There wasn't enough time to redesign and install a larger shaft, so we had to leave the counterweight out entirely. Luckily, the system had enough power to work all right without the counterweight Experience like that has convinced me that generally it's better to overdesign than underdesign. So, on this new machine I chose a natural frequency above thirty cycles even though no shake table testing is required."

By making the guide shaft stiff enough to have a natural frequency of around 50 cps, Mr. Howard expected that there would be negligible deflection. Hence, he made no analysis of bending strength or deflection. He thought sufficient clearance between the counterweight can and the stationary frame would be 1/8 to 1/4 inch.

To suspend the counterweight he planned to use a steel cable and "shank ball," selected from a cable catalog, excerpts of which appear in Exhibit 5. "A safety factor of around 5 seems reasonable to me," he said "That is a figure I've used before with success, and is in the range I would expect to find other engineers using. I think the figure first came to my attention in a magazine article some years ago which said that for shipment by truck it was advisable to protect against a maximum loading of about 5 G's. That figure may also have appeared in some literature put out by shock mount makers."

"But the cable should also be as flexible as possible so I don't need an enormous pulley to run it over Bending a heavy cable over a small pulley can make it fail in fatigue. So it's better not to use a cable heavier than necessary and it's better to use one made of smaller strands.

"Now my questions are what dimensions to make the can, what size and type of cable to hang it with, and where to hook it onto the moving frame, whether off center someplace or in the middle I also need to choose the vertical support bar diameter. Then I can design the frame for horizontal movement, after which come the horizontal supports and main frame "

EXHIBIT 1 - Mr. Howard's Early Sketch

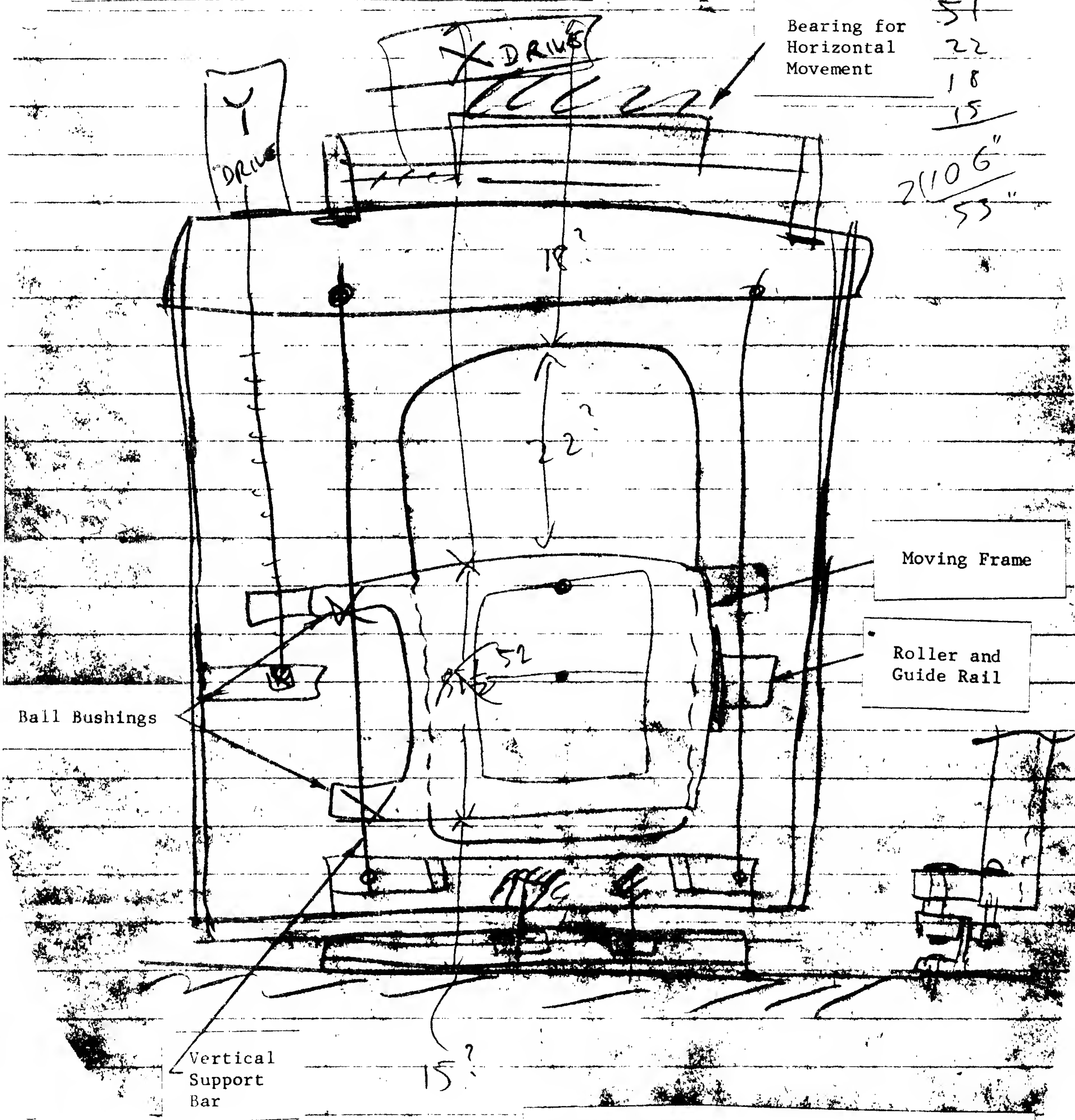

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Exhibit 2
Excerpt From Catalog

BALL BUSHING ENGINEERING SPECIFICATIONS

Table #2 — PRECISION Series A — Dimensions & Load Ratings

Series A Ball Bushing Number	Working Bore A Inches	A Tol. +.0000 to	Outside Diameter B Inches	B Tol. +.0000 to	Length C Inches	C Tol. +.000 to	Dist. Between Retaining Rings D Inches	D Tol	Maximum Permissible Shaft Dia*** Inches	Recommended Housing Bore Norm. Fit Inches	Tol. .0000 to	Press Fit Inches	Tol. .0000 to	Ball Diameter Inches	Number of Ball Circuits	Bushing Weight Pounds	Rated Load* Static Pounds	Rated Load* Rolling** Pounds	Series A Ball Bushing Number
A-4812	.2500	[illegible]	.5000	.0003	.750	.015	.437	±.010	.2490	.5000	+.0005	.4990	+.0005	1/16"	3	.02	22	13	A-4812
A-61014	.3750	[illegible]	.6250	.0003	.875	.015	.562	±.010	.3740	.6250	+.0005	.6240	+.0005	1/16"	4	.06	38	21	A-61014
A-81420	.5000	[illegible]	.8750	.0003	1.250	.015	.875	±.010	.4990	.8750	+.0005	.8740	+.0005	3/32"	4	.08	72	46	A-81420
A-122026	.7500	.0005	1.2500	.0003	1.625	.015	1.062	±.010	.7490	1.2500	+.0005	1.2490	+.0005	1/8"	5	.21	162	109	A-122026
A-162536	1.0000	[illegible]	1.5625	.0003	2.250	.015	1.625	±.010	.9990	1.5625	+.0005	1.5615	+.0005	5/32"	5	.38	262	202	A-162536
A-203242	1.2500	[illegible]	2.0000	.0005	2.625	.020	1.875	±.015	1.2490	2.0000	+.0010	1.9983	+.0010	3/16"	6	1.10	465	344	A-203242
A-243848	1.5000	[illegible]	2.3750	.0005	3.000	.020	2.250	±.015	1.4989	2.3750	+.0010	2.3733	+.0010	7/32"	6	1.43	695	535	A-243848
A-324864	2.0000	.0008	3.0000	.0006	4.000	.020	3.000	±.015	1.9987	3.0000	+.0010	2.9982	+.0010	1/4"	6	2.75	1100	850	A-324864
A-406080	2.5000	.0010	3.7500	.0008	5.000	.025	3.750	±.015	2.4985	3.7500	+.0010	Not Normally Recommended		5/16"	6	5.50	1710	1380	A-406080
A-487296	3.0000	.0012	4.5000	.0010	6.000	.030	4.500	±.015	2.9983	4.5000	+.0010			3/8"	6	9.50	2460	2000	A-487296
A-6496128	4.0000	[illegible]	6.0000	.0012	8.000	.040	6.000	±.020	3.9976	6.0000	+.0010			1/2"	6	20.20	4400	3800	A-6496128

*Based on a shaft hardness of Rockwell 60C
**Based on a travel life of 2 million inches. (See Page 25)
***For normal fit slightly larger shafts may be used with caution. (See Page 18)

STAINLESS STEEL Series A, XA, ADJ and OPN BALL BUSHINGS are available made entirely of Stainless Steel. They are identified by the suffix SS following the part number (Example XA 81420 SS). Series A and XA are stocked only in sizes up to and including 1". Series ADJ and OPN are stocked in ½", ¾" and 1". *For larger sizes see note on Page 29.*

Table #3 — SUPER PRECISION Series XA — Dimensions & Load Ratings

Series XA Ball Bushing Number	Working Bore A Inches	A Tol. +.0000 to	Concentricity T.I.R.	Outside Diameter B Inches	B Tol. +.000 to	Length C Inches	C Tol. +.000 to	Dist. Between Retaining Rings D Inches	D Tol	Maximum Permissible Shaft Dia*** Inches	Recommended Housing Bore Norm. Fit† Inches	Tol. .0000 to	Press Fit Inches	Ball Diameter Inches	Number of Ball Circuits	Bushing Weight Pounds	Rated Load* Static Pounds	Rated Load* Rolling** Pounds	Series XA Ball Bushing Number
XA-4812	.2500	.0003	[illegible]	.5000	.0004	.750	.015	.437	±.010	.2495	.5000	+.0005	NOT NORMALLY RECOMMENDED	1/16"	3	.02	22	13	XA-4812
XA-61014	.3750	.0003	.0005	.6250	.0004	.875	.015	.562	±.010	.3745	.6250	+.0005		1/16"	4	.06	38	21	XA-61014
XA-81420	.5000	.0003	.0005	.8750	.0004	1.250	.015	.875	±.010	.4995	.8750	+.0005		3/32"	4	.08	72	46	XA-81420
XA-122026	.7500	.0003	.0005	1.2500	.0004	1.625	.015	1.062	±.010	.7495	1.2500	+.0005		1/8"	5	.21	162	109	XA-122026
XA-162536	1.0000	.0003	[illegible]	1.5625	.0004	2.250	.015	1.625	±.010	.9995	1.5625	+.0005		5/32"	5	.38	262	202	XA-162536
XA-203242	1.2500	.0004	[illegible]	2.0000	.0005	2.625	.020	1.875	±.015	1.2495	2.0000	+.0010		3/16"	6	1.10	465	344	XA-203242
XA-243848	1.5000	.0004	[illegible]	2.3750	.0005	3.000	.020	2.250	±.015	[illegible]	2.3750	+.0010		7/32"	6	1.43	695	535	XA-243848
XA-324864	2.0000	.0004	.0010	3.0000	.0006	4.000	.020	3.000	±.015	[illegible]	3.0000	+.0010		1/4"	6	2.75	1100	850	XA-324864
XA-406080	2.5000	.0005	.0015	3.7500	.0008	5.000	.025	3.750	±.015	2.4993	3.7500	+.0010		5/16"	6	5.50	1710	1380	XA-406080
XA-487296	3.0000	.0006	.0015	4.5000	.0010	6.000	.030	4.500	±.015	2.9992	4.5000	+.0010		3/8"	6	9.50	2460	2000	XA-487296
XA-6496128	4.0000	.0010	.0020	6.0000	.0012	8.000	.040	6.000	±.020	3.9988	6.0000	+.0010		1/2"	6	20.20	4400	3800	XA-6496128

*Based on a shaft hardness of Rockwell 60C
**Based on a travel life of 2 million inches. (See Page 25)
†For extreme precision, tolerance may be reduced.
***Slightly larger shafts may be used with caution. (See Page 18)

Table #4 — COMMERCIAL GRADE Series B — Dimensions & Load Ratings

Sold Only in lots of 250 or more.

Series B Ball Bushing Number	Working Bore A Inches	A Tol. .0000 to	Outside Diameter* B Inches	B Tol. +.0000 to	Length C Inches	C Tol. +.000 to	Dist. Between Retaining Rings D Inches	D Tol.	Maximum Permissible Shaft Diameter Norm. Fit Inches	Maximum Permissible Shaft Diameter Press Fit Inches	Recommended Housing Bore Norm. Fit Inches	Tol. .0000 to	Press Fit† Inches	Tol. .0000 to	Ball Diameter Inches	Number of Ball Circuits	Bushing Weight Pounds	Rated Load** Static Pounds	Rated Load** Rolling*** Pounds	Series B Ball Bushing Number
B-4812	.2500	[illegible]	.5000	.0015	.750	.020	.437	±.010	.2495	[illegible]	.5000	+.0010	.4980	+.0005	1/16"	3	.02	19	11	B-4812
B-61014	.3750	[illegible]	.6250	.0015	.875	.020	.562	±.010	.3745	.3740	.6250	+.0010	.6230	+.0005	1/16"	4	.06	33	18	B-61014
B-81420	.5000	[illegible]	.8750	.0015	1.250	.020	.875	±.010	.4995	.4990	.8750	+.0010	.8730	+.0005	3/32"	4	.08	61	39	B-81420
B-122026	.7500	[illegible]	1.2500	.0020	1.625	.025	1.062	±.015	.7495	[illegible]	1.2500	+.0010	1.2475	+.0005	1/8"	5	.21	138	93	B-122026
B-162536	1.0000	[illegible]	1.5625	.0020	2.250	.025	1.625	±.015	[illegible]	.9990	1.5625	+.0010	1.5600	+.0005	5/32"	5	.37	222	172	B-162536
B-203242	1.2500	[illegible]	2.0000	.0020	2.625	.030	1.875	±.015	1.2495	[illegible]	2.0000	+.0010	1.9970	+.0010	3/16"	6	1.10	400	292	B-203242
B-243848	1.5000	[illegible]	2.3750	.0020	3.000	.030	2.250	±.015	1.4994	[illegible]	2.3750	+.0010	2.3720	+.0010	7/32"	6	1.40	590	455	B-243848

*Slight out-of roundness may result from the heat treatment of Series B bearings, making it difficult to measure the true O.D. The bearing will return substantially to its original roundness when inserted into the recommended housing bore for either normal or press fit.
†Do not press fit in soft metal housings. Use normal fit to avoid shearing.
**Based on a shaft hardness of Rockwell 60C
***Based on a travel life of 2 million inches. (See Page 25)

COEFFICIENT of FRICTION

Low friction is an important feature of Ball Bushings.

The coefficient of friction of BALL BUSHINGS 0.001 to 0.004 is extremely low, and is approximately that of *radial* ball bearings. It is far less than the coefficient of friction of sliding surfaces and, more important, *is far more constant.* This applies to the coefficient of rolling or operating friction as well as to the coefficient of static or break-away friction. From a practical standpoint, this low coefficient of friction is minute enough to be disregarded in most applications. A *constant* low coefficient of friction is imperative in many applications where stick-slip action impairs performance.

Extensive testing under controlled conditions has determined coefficients of rolling and static friction for the entire series of BALL BUSHINGS. These values may be used to estimate forces required to overcome frictional resistance in specific applications.

The formula used to determine frictional resistance during operation is:

$P = L \times f_r$, where P = Frictional resistance (pounds)
L = Applied load (pounds), perpendicular to centerline of shaft
f_r = Coefficient of rolling friction

The formula used to determine static *frictional resistance is:*

$P = L \times f_o$, where f_o = Coefficient of static friction

Following is a list of coefficients of rolling friction (fr) of BALL BUSHINGS operating on hardened and ground shafts of recommended diameters. These values are grouped according to the number of ball circuits in each Bushing as friction coefficients are constant among Bushings having 3 and 4 ball circuits but slightly less for Bushings with 5 or 6 ball circuits. To make the table easy to use Bushing sizes in each group are also listed.

The values for the coefficient of static or break-away friction are also listed. These values are not measurably affected by the number of ball circuits in the Bushing or by conditions of lubrication.

Variables affecting friction of Ball Bushings

SPEED There are no appreciable variations in coefficients of rolling friction at various speeds.

LUBRICATION Dry BALL BUSHINGS have the lowest coefficient of friction due to the complete absence of lubricant surface tension (meniscus drag) effects. Values for grease lubrication range from 100% greater in the smaller sizes to 20%-to-50% greater in the larger sizes. Oil lubrication (medium-heavy, viscosity 64 c.s. @ 100° F.) gives frictional values slightly higher than those for grease lubrication.

SEAL FRICTION - Where seals or flexible bellows are used to retain lubricant or to prevent entry of foreign particles, frictional resistance from these elements must also be taken into consideration in calculations made to determine total frictional drag. Where minimum friction is desired, sealing elements with minimum frictional drag are used or sealing elements are eliminated by design. BALL BUSHINGS are not as critical as radial ball bearings with respect to foreign particles as they are pushed aside by the ball action and are not trapped between balls and ball conforming grooves.

Coefficients of Rolling Friction (f_r) of Ball Bushings

$$f_r = \frac{P}{L}$$ where P equals *frictional resistance* and L equals *applied load*

BUSHING I D	NUMBER of BALL CIRCUITS	CONDITION of LUBRICATION	LOAD IN % OF ROLLING LOAD RATING (for 2,000,000 inches of travel life)				
			125%	100%	75%	50%	25%
¼", ⅜", ½"	3 & 4	No Lube	0011	0011	.0012	.0016	.0025
		Grease Lube	.0019	0021	0024	.0029	0044
		Oil Lube	0022	.0023	.0027	0032	0045
¾", 1"	5	No Lube	0011	.0011	0012	0015	.0022
		Grease Lube	0018	0019	0021	0024	.0033
		Oil Lube	0020	0021	0023	0027	0036
1¼" thru 4"	6	No Lube	.0011	.0011	0012	.0014	.0019
		Grease Lube	0016	.0016	.0017	.0018	.0022
		Oil Lube	0018	0018	0019	0021	.0027

Coefficients of Static Friction (f_o) of Ball Bushings

LOAD IN % OF ROLLING LOAD RATING				
125%	100%	75%	50%	25%
0028	0030	.0033	0036	0040

Values are based on use of shafts of recommended diameters, hardened to Rockwell 58-63C.

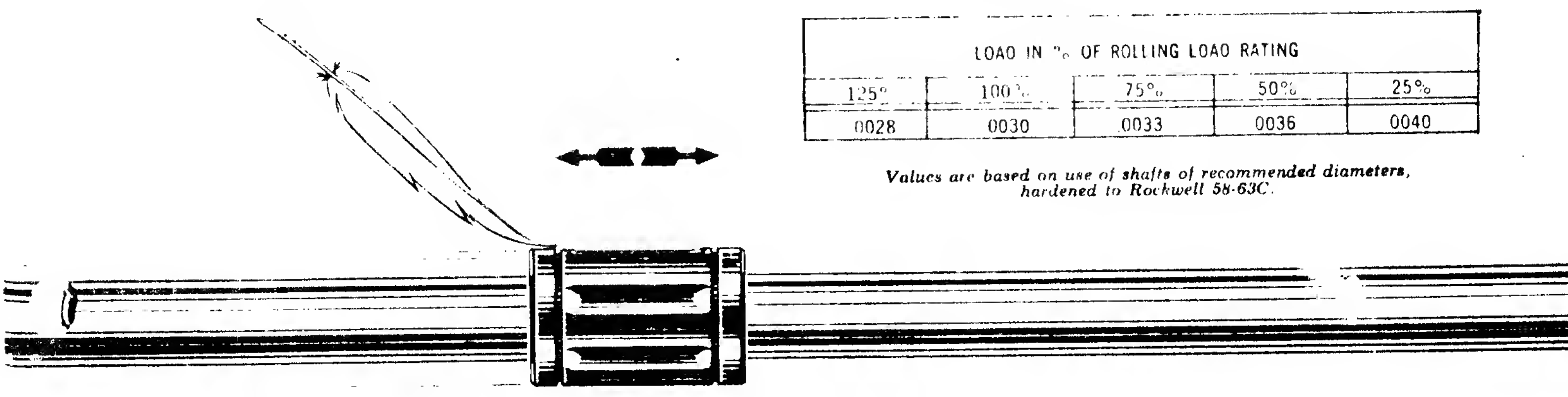
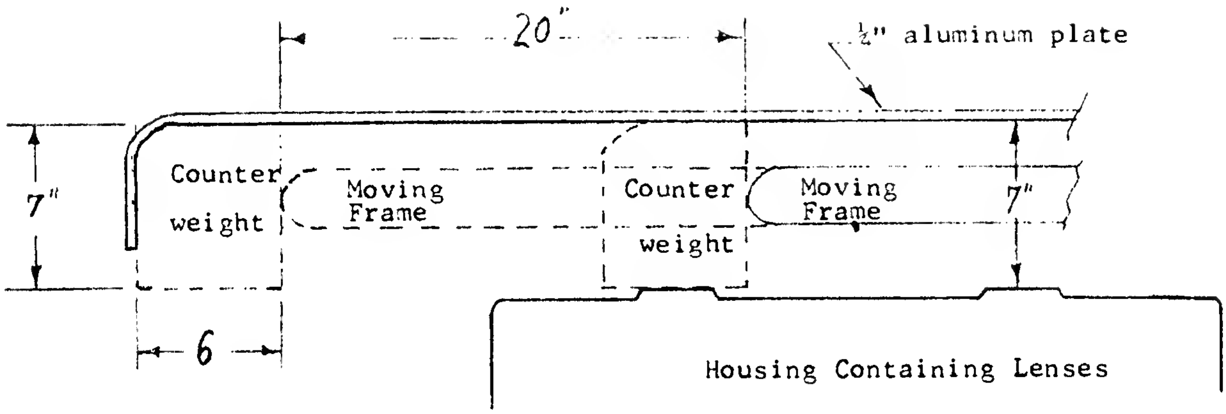


Exhibit 3 - Location of Counterweight



Top View

Positions of the frame and counterweight which move both vertically and horizontally are shown in two positions, one at the left limit of travel and the other at the right limit of travel.

Exhibit 4 - Calculations from Mr. Howard's Notebook

6/12/63

WT. OF VERT. MOVING BALCONY

(see 3/28, ETC)

2 PCS 20x20 GLASS	18 LBS
2 FRAMES AROUND GLASS (STL)	15 LBS
2 BALL BUSHINGS	11 LBS
ROLLERS & BKTS, ETC.	11 LBS
MAIN CASTING FRAME (ALUM)	112 LBS
Amount to be Counterbalanced	167 LBS

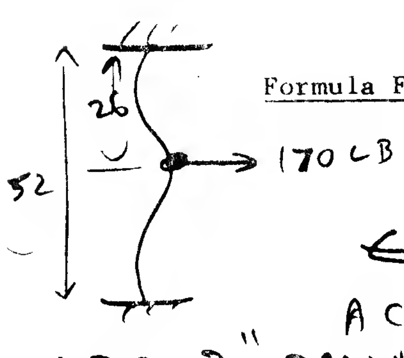
$$C'\text{WT. VOL.} = \frac{\left(\frac{167\ \text{LB}}{.41\ \text{LB/IN}^3}\right)}{.7\ \text{SPACE FACTOR}} = 600\ \text{in}^3$$

Counterweight Volume

IF WE USE SPLINED SHAFT — 52" LONG, 170 LB C'W

Counterweight Guide Shaft Frequency Determination

Assume 2" SHAFT — HAS 1.66 ROOT DIA.



Formula From Magazine

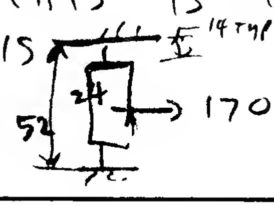
$$f_n = 43.3\sqrt{\frac{3\times10^7\times5\times(1.66)^4}{170\times10^2\times52^3}} = 43.3\sqrt{\frac{15\times1.66^4}{1.7\times(5.2)^3}}$$

$$= 43.3\sqrt{.477} = (43.3)(.691) = 29.9\ \text{CPS, SAY 30 CPS}$$

— HOWEVER —

THIS IS TOO SIMPLE A REPRESENTATION —

ACTUALLY



Coeff for = 70, not 43.3

So, we should have > 30 CPS

USE 2" SPLINE (MIN.)

USE 2½" SPLINE (IF POSS.)

Exhibit 4 - Continued

WITH $2\frac{1}{2}$ DIA. SPLINE (ROOT D = 2.1)

$$f_m = 43.3\sqrt{.477\left(\frac{2.1}{1.66}\right)^4} = 43.3\sqrt{1.22} = 43.3 \times 1.105 = \underline{\underline{47.8\text{ CPS}}}$$

(SIMPLIFIED CALC.)

MW.

IF WE HAVE LARGER COEFFIC. (WHICH WE DO) — WILL BE > 47.8 CPS

coeff should be at least 50 —

giving $f_m = 50 \times 1.105 = \underline{55.2\text{ CPS}}$

Make A $2\frac{1}{2}$ DIA. Spline — TRY

Exhibit 5 - Cable Catalog Excerpt

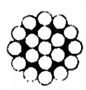
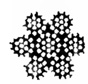
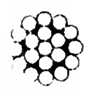
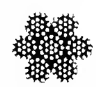
SPECIFICATIONS

WIRE ROPE FOR ASSEMBLIES

Tables below cover Preformed galvanized steel and stainless steel most generally used for wire rope assemblies.

PREFORMED

Breaking Strength in POUNDS | **Breaking Strength in POUNDS**

DIAMETER IN INCHES	GALVANIZED STEEL			STAINLESS STEEL			DIAMETER IN INCHES
	MIL-C-6940	MIL-C-1511	MIL-C-1511	MIL-C-5693A	MIL-C-5424	MIL-C-5424	
	 1 x 19 ** ††	 7 x 7	 7 x 19	 1 x 19 **	 7 x 7 †	 7 x 19 *	
1/16	500	480		500	480		1/16
3/32	1200	920	1000	1200	920	920	3/32
1/8	2100	1700	2000	2100	1700	1760	1/8
5/32	3300	2600	2800	3300	2400	2400	5/32
3/16	4700	3700	4200	4700	3700	3700	3/16
7/32	6300	4800	5600	6300	4800	5000	7/32
1/4	8200	6100	7000	8200	6100	6400	1/4
9/32	10300	7600	8000	10300	7600	7800	9/32
5/16	12500	9200	9800	12500	9000	9000	5/16
3/8	17500	13100	14400	17500	12000	12000	3/8
7/16	23400		17600	23400		16300	7/16
1/2	30300		22800	29700		22800	1/2
9/16			28500			28500	9/16
5/8			35000			35000	5/8
3/4			49600			49600	3/4
7/8			66500			66500	7/8
1			85400			85400	1

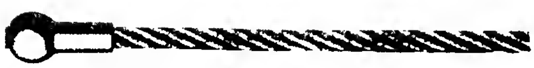
†[illegible]" and larger made 6 x 7 with I.W.R.C.
*[illegible]" and larger made 6 x 19 with I.W.R.C.

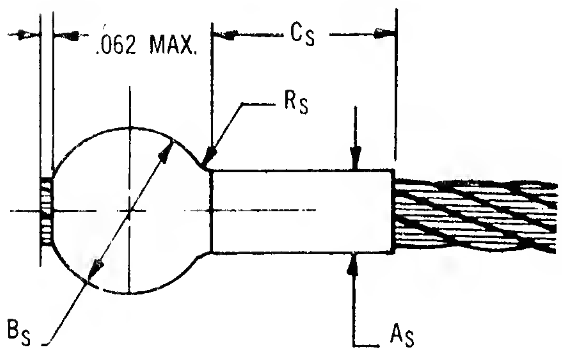
**Strength efficiency references in this catalog are based on the use of wire rope, not strand. For efficiences of swaged terminals when used with strand constructions (1x19) refer to Sales Department, Wilkes-Barre.

MS 20664

Exhibit 5 - Continued

INDUSTRIAL STANDARDS
WIRE ROPE ASSEMBLIES





MATERIAL:

Corrosion Resisting Stainless Steel.

STRENGTH:

These terminals will develop the rated breaking strength of the wire rope.

MS-20664 (see first column for Dash Part Number)

DIMENSIONS AFTER SWAGING

MS DASH PART NO.	NOM. CABLE DIA.	MIN. ALLOW. BREAK. STR. (POUNDS) AIRC. CABLE	As		Bs		Cs	Rs
COR. RES. STEEL			DIA.	TOL.	SPH. DIA.	TOL.	REF.	MAX. RAD.
C2	1/16	480	.112	+.000 −.003	.190	+.000 −.003	.156	.014
C3	3/32	920	.143	+.000 −.003	.253	+.000 −.003	.234	.019
C4	1/8	2000	.190	+.000 −.003	.315	+.000 −.003	.313	.023
C5	5/32	2800	.222	+.000 −.004	.379	+.000 −.004	.391	.028
C6	3/16	4200	.255	+.000 −.005	.442	+.000 −.005	.469	.033
C7	7/32	5600	.302	+.000 −.005	.505	+.000 −.005	.547	038
C8	1/4	7000	.348	+.000 −.005	.567	+.000 −.005	.625	.042
C9	9/32	8000	.382	+.000 −.007	.632	+.000 −.007	.750	.046
C10	5/16	9800	.413	+.000 −.007	.694	+.000 −.007	.813	.046

INTERCHANGEABILITY RELATIONSHIP: MS-20663 & AN-663 PARTS IDENTIFIED BY THE SAME DASH NUMBER ARE UNIVERSALLY, FUNCTIONALLY, & DIMENSIONALLY INTERCHANGEABLE.

ALLOWABLE VARIATION: ±.005 ON ALL DIMENSIONS UNLESS OTHERWISE SPECIFIED   GLES ±3°

MFG. IN ACCORDANCE WITH MIL-T-781

CABLE IN ACCORDANCE WITH MIL-W-1511 & MIL-C-5424

Courtesy of American Chain and Cable Company